Computer Modeling of a Single-Stage Lithium Bromide/Water Absorption Refrigeration Unit

F. L. Lansing
DSN Engineering Section

Solar-assisted refrigeration systems have been proposed for application in Goldstone energy conservation projects. This article describes and analyzes the dynamic simulation and the computer modeling procedure of one of these systems, namely a lithium bromide/water absorption refrigeration system. A new analytical expression that fits the three-dimensional surface of LiBr concentration, refrigerant temperature and solution temperature in the range of interest from 0.50 to 0.65 kg LiBr/kg solution is presented with a standard deviation of ±0.2 percent. This will save considerable computing time and effort required for evaluation of system performance. A numerical example from typical running conditions is added to show the relative weight of each parameter used together with the sequence of program steps followed. The results from this simulation are heat rates, line concentrations, pressures and the overall coefficient of performance.

I. Introduction

In recent years, the lithium bromide/water absorption system has become prominent in refrigeration for air conditioning. It possesses several advantages over the other types of absorption systems, such as:

(1) It has the highest coefficient of performance (COP) compared to other single-stage absorption units at the same cycle temperatures.

(2) It is composed of simpler components since it can work efficiently without the need of rectification columns. A basic generator is sufficient due to the nonvolatility of the absorbent (LiBr), allowing only water vapor to be driven off the generator.

(3) Less pump work is needed compared to other units due to operation at vacuum pressures.

On the other hand, the lithium bromide/water absorption system has some drawbacks such as:

(1) It is limited to relatively high evaporating temperatures since the refrigerant is water. This means that evaporation temperature above 0°C must generally be satisfied to prevent flow freezing.

(2) Crystallization of LiBr salt at moderate concentrations ($>$0.65 kg LiBr/Kg solution) will tip off the cycle range of operation.

(3) The systems have to be designed in hermetically sealed units since they operate at vacuum pressures. Improper operation would result if leakage of air into the system occurred.

Irrespective of its drawbacks, the LiBr/water unit is still considered the most economical for this kind of refrigeration technique. It has been selected as a candidate refrigeration system in connection with a proposed solar assisted equipment for application in Goldstone energy conservation projects.

This article describes and analyzes the computer modeling of such units. The modeling procedure is generalized to enable those concerned with use or evaluation of cycles employing this material to save considerable time and effort required for calculations.

II. Thermodynamic Cycle

The system components and the working fluid states are shown in Figs. 1 and 2. There are four basic heat exchanger surfaces: the evaporator, the absorber, the generator, and the condenser, in addition to a liquid-liquid heat exchanger. Lithium bromide is, basically, nothing more than salt water. However, lithium bromide is a salt with an especially strong attraction for water.

The cycle of operation may be started as shown in Fig. 1, from the evaporator. The refrigerant (water) is evaporated while it is taking heat from the fluid being chilled (air for instance). The water vapor (state 10) is then sucked up by lithium bromide spray injected into the absorber, thus the name absorption system. Due to the exothermic reaction taking place in the absorption process, heat has to be removed, and the mixture of lithium bromide and refrigerant vapor at this stage is called the "strong solution" (state 1). "Strong" and "weak" refer to the amount of refrigerant present. The strong solution is then pumped (state 2) through a liquid-liquid heat exchanger (state 3) to the generator. This heat exchanger will improve the cycle performance, as will be shown later. In the generator (sometimes called the concentrator) the strong solution is heated and boiled by an external heat source to release the refrigerant vapor (state 7), leaving behind a concentrated LiBr/water solution (state 4). The latter is called "weak solution" since

it contains a smaller amount of refrigerant. The refrigerant vapor leaving the generator is condensed (state 8) in the condenser and is directed to the evaporator through an expansion valve (state 9). The weak solution flows back to the absorber through the liquid-liquid heat exchanger as a spray (state 6) to complete the cycle.

The thermodynamic cycle is sketched on the equilibrium temperature-pressure-concentration chart as shown in Fig. 2. It is bounded by two constant concentration lines: X_1 and X_4 for the absorber and generator concentration, respectively, and two constant pressure lines: P_e and P_c for the evaporator and condenser pressures, respectively.

For an efficient air conditioning application, the evaporator temperature t_e should be low enough to dehumidify the air. In practice, it ranges from 4.5 to 10°C according to the load conditions. The heat rejection temperatures t_a and t_c for the absorber and the condenser, respectively, vary according to the type of cooling medium (air, water), ambient conditions of dry and wet bulb temperatures, and whether their cooling lines are connected in series or parallel with each other.

The generator temperature t_g depends on the source of heat available, i.e., solar, gas or steam. However, a minimum temperature of 80°C should be maintained to provide efficient operation.

The operational function of the liquid-liquid heat exchanger in the cycle will be the reduction of the weak solution temperature t_4, leaving the generator and increasing the strong solution temperature t_1 leaving the absorber. The result is that it reduces the heat input necessary to the generator while reducing the heat rejected from the absorber by heat exchanging between the two solutions. This heat exchanger does increase the coefficient of performance of the system and is, therefore, always desirable.

III. Thermodynamic Properties

Enthalpies of the water (refrigerant) and LiBr (absorbent) solutions were calculated with reference temperature at 25°C (Refs. 1–4). The following expressions were found essential to the calculation of the cycle performance:

(1) The enthalpy of pure water liquid at temperature t°C

$$= (t - 25) \quad \text{kcal/kg} \quad (1)$$

(2) The enthalpy of saturated water vapor at temperature $t°C$

$$= (572.8 + 0.417\,t) \quad \text{kcal/kg} \tag{2}$$

(latent heat $= 597.8 - 0.583\,t \pm 0.4\,\text{kcal/kg}$)

(3) The enthalpy of superheated steam at temperature $t°C$ and at a pressure equal to the saturation pressure of steam at temperature $t_s°C$

$$= (572.8 + 0.417\,t_s) + 0.46\,(t - t_s)$$
$$= 572.8 + 0.46\,t - 0.043\,t_s \quad \text{kcal/kg} \tag{3}$$

by taking the specific heat of water vapor $= 0.46$ kcal/kg°C at very low pressures (0.01 to 0.1 atm).

(4) The specific heat of lithium bromide/water solution of concentration X kg LiBr/kg solution is given by

$$C_x = 1.01 - 1.23\,X + 0.48\,X^2 \quad \begin{array}{l}\text{kcal/kg}\\\text{solution}\end{array} \tag{4}$$

(5) The enthalpy of LiBr/water solution of concentration X kg LiBr/kg solution at 25°C is

$$H_{X,25} = 68.06 - 456.67\,X + 416.67\,X^2 \quad \begin{array}{l}\text{kcal/kg}\\\text{solution}\end{array} \tag{5}$$

(6) The enthalpy of LiBr/water solution of concentration X kg LiBr/kg solution at temperature $t°C$ is given by

$$H_{X,25} + C_X(t - 25)$$

or

$$= (42.81 - 425.92\,X + 404.67\,X^2)$$
$$+ (1.01 - 1.23\,X + 0.48\,X^2)\,(t) \tag{6}$$

(7) In the range of concentration from 0.50–0.65 kg LiBr/kg solution, the author found that it is possible to fit the refrigerant temperature t_R, the saturated solution temperature t_m, and the concentration X by the relation

$$(t_R\,°C) = (49.04 - 134.65\,X)$$
$$+ (1.125 - 0.47\,X)\,(t_m\,°C)$$

with a standard deviation $\pm 0.2\%$. This may be rewritten as

$$X = \frac{49.04 + 1.125\,t_m - t_R}{134.65 + 0.47\,t_m} \tag{7}$$

i.e., X is completely defined by the set $\{t_R, t_m\}$.

(8) The saturated vapor pressure P in mm Hg corresponding to saturation temperature $T°K$ for pure water is given by (Ref. 5)

$$\log_{10} P \text{ mm Hg} = 7.8553 - \frac{1555}{T} - \frac{11.2414 \times 10^4}{T^2} \tag{8}$$

IV. Performance Calculations

The determination of the thermodynamic properties of each state in the cycle, the amount of heat transfer in each component, and the flow rates at different lines depend on the following set of input parameters:

Generator temperature t_g, °C

Evaporator temperature t_e, °C

Condenser temperature t_c, °C

Absorber temperature t_a, °C

Liquid-liquid heat exchanger effectiveness E_L

Refrigeration load Q_E, tons

The above set can be determined from actual running measurements or assumed by a first reasonable estimate to cycle performance.

Together with the assumptions of neglecting the pump-work and neglecting the pressure drop in components and lines and assigning saturation conditions to states numbers 1, 4, 8, and 10 in Fig. 1, the properties are determined as follows:

A. Absorber Concentration

This is determined by Eq. (7) using t_a for the solution temperature and t_e for the water temperature corresponding to the evaporator pressure P_e:

$$X_1 = X_2 = X_3 = X\,\substack{strong\\solution} = \frac{49.04 + 1.125\,t_a - t_e}{134.65 + 0.47\,t_a}$$

$$\text{kg LiBr/kg solution} \tag{9}$$

B. Generator Concentration

This is evaluated, from Eq. (7), using t_g for the solution temperature and t_c for the refrigerant temperature corresponding to the condenser pressure P_c

$$X_4 = X_5 = X_6 = X \underset{\text{solution}}{weak} = \frac{49.04 + 1.125\,t_g - t_c}{134.65 + 0.47\,t_g}$$

$$\text{kg LiBr/kg solution} \qquad (10)$$

It may be noted that X_4 is always larger than X_1, and

$$X_7 = X_8 = X_9 = X_{10} = \text{zero} \qquad \text{kg LiBr/kg solution} \qquad (11)$$

for the pure water flow in the condenser and evaporator.

C. Pressure Limits in the Cycle

Using Eq. (8), it is possible to evaluate the pressure in every line as follows:

$P_{evaporator}, P_e = P_1 = P_6 = P_9 = P_{10}$ in mm Hg is given by

$$\log_{10} P_e = 7.8553 - \frac{1555}{t_e + 273.15} - \frac{11.2414 \times 10^4}{(t_e + 273.15)^2} \qquad (12)$$

and the condenser pressure, $P_c = P_2 = P_3 = P_4 = P_5 = P_7 = P_8$ in mm Hg is given by

$$\log_{10} P_c = 7.8553 - \frac{1555}{(t_c + 273.15)} - \frac{11.2414 \times 10^4}{(t_c + 273.15)^2} \qquad (13)$$

D. Flow Rates

Enthalpy of saturated liquid water, state 8, is given by Eq. (1) at the condenser temperature t_c as

$$H_8 = (t_c - 25) \qquad \text{kcal/kg} \qquad (14)$$

The throttling processes from 8 to 9 and that from 5 to 6 give

$$\left.\begin{array}{l} H_8 = H_9 \\ H_5 = H_6 \end{array}\right\} \qquad (15)$$

Enthalpy of saturated water vapor, state 10, is given by Eq. (2) at the evaporator temperature t_e as

$$H_{10} = 572.8 + 0.417\,t_e \qquad (16)$$

Applying the first law of thermodynamics to the evaporator will give

$$Q_E = m_R (H_{10} - H_9)$$

where m_R is the refrigerant flow rate, equals the difference between the strong and weak solution rates. By using Eq. (15)

$$m_R = \frac{Q_E}{H_{10} - H_8} \qquad (17)$$

On the other hand, the lithium bromide mass balance in the absorber gives

$$m_w X_6 + m_R X_{10} = m_s X_1 = (m_w + m_R)X_1$$

and by using Eqs. (11) and (17), then

$$m_w = \left(\frac{Q_E}{H_{10} - H_8}\right)\left(\frac{X_1}{X_4 - X_1}\right), \qquad (18)$$

$$m_s = \frac{Q_E}{(H_{10} - H_8)}\left(\frac{X_4}{X_4 - X_1}\right) \qquad (19)$$

Since the concentrations X_1 and X_4 are restricted not to exceed certain limits to avoid crystallization problems, and if the temperatures of the cycle are set to vary according to ambient and load conditions, the mass flow rates in the different lines will be varied accordingly. This necessitates the existence of LiBr and water solution inventories to be used for flow compensation, especially at times when variations of load, hot water temperature, and cooling water temperature do occur.

E. Liquid-Liquid Heat Exchanger Temperatures

Once the heat exchanger effectiveness E_L, the mass flow rates (m_w, m_s) and the concentrations (X_1, X_4) are given, it is possible to determine the solution temperatures t_3 and t_5 from Fig. 3 as follows:

Based on the weak solution side, which has the minimum heat capacity, the effectiveness E_L is defined by (Ref. 6)

$$E_L = \frac{t_g - t_5}{t_g - t_a} \qquad (20a)$$

or based on the strong solution side

$$E_L = \frac{(m_s \cdot C_{X1}) \cdot (t_3 - t_a)}{(m_w \cdot C_{X4}) \cdot (t_g - t_a)} \qquad (21a)$$

where C_{X1} is specific heat of the strong solution whose concentration is X_1, and C_{X4} is the specific heat of the weak solution whose concentration is X_4. Both C_{X1}, C_{X4} are determined from Eq. (4) as

$$C_{X1} = 1.01 - 1.23\,X_1 + 0.48\,X_1^2 \left.\right\}$$
$$C_{X4} = 1.01 - 1.23\,X_4 + 0.48\,X_4^2 \left.\right\} \tag{22}$$

Equations (20) and (21) are rewritten using Eqs. (18) and (19) to give the temperatures t_3 and t_5 as:

$$t_5 = t_g - E_L(t_g - t_a) \tag{20b}$$

and

$$t_3 = t_a + \left[E_L \cdot \left(\frac{X_1}{X_4} \right)\left(\frac{C_{X4}}{C_{X1}} \right)(t_g - t_a) \right] \tag{21b}$$

The enthalpies H_1 and H_5 are then calculated using Eq. (6) as follows:

$$H_1 = (42.81 - 425.92\,X_1 + 404.67\,X_1^2)$$
$$+ (1.01 - 1.23\,X_1 + 0.48\,X_1^2) \cdot (t_a) \tag{23a}$$

$$H_5 = (42.81 - 425.92\,X_4 + 404.67\,X_4^2)$$
$$+ (1.01 - 1.23\,X_4 + 0.48\,X_4^2) \cdot (t_5) \tag{23b}$$

F. Heat Transfer in Condenser, Generator, and Absorber

The enthalpy of water vapor leaving the generator and entering the condenser (state 7) is determined by Eq. (3) as

$$H_7 = 572.8 + 0.46\,t_g - 0.043\,t_c \tag{24}$$

The heat balance of the condenser gives

$$Q_C = m_R(H_7 - H_8) \tag{25a}$$

or using Eq. (17) for m_R expression, it becomes

$$Q_C = \frac{Q_E}{(H_{10} - H_8)} \cdot (H_7 - H_8) \tag{25b}$$

Heat balance for the combined generator and heat exchanger control volume gives

$$Q_G = m_w H_5 + m_R H_7 - m_s H_2 \tag{26}$$

Since the pumpwork is negligible, then

$$H_1 \simeq H_2 \tag{27}$$

Using Eqs. (17), (18), (19), and (27), it is possible to write Q_G as

$$Q_G = \frac{Q_E}{(H_{10} - H_8)}\left[\frac{X_1 H_5}{(X_4 - X_1)} + H_7 - \frac{X_4 H_1}{(X_4 - X_1)} \right] \tag{28}$$

Heat balance of the absorber gives Q_A as

$$Q_A = m_w H_6 + m_R H_{10} - m_s H_1$$

Using Eqs. (15), (17), (18), and (19), Q_A is rewritten as

$$Q_A = \frac{Q_E}{(H_{10} - H_8)}\left[\frac{X_1 H_5}{(X_4 - X_1)} + H_{10} - \frac{X_4 H_1}{(X_4 - X_1)} \right] \tag{29}$$

Equations (25b) and (29) are governed by the first law of thermodynamics in the form

$$Q_C + Q_A = Q_G + Q_E \tag{30}$$

G. Coefficient of Performance (COP)

This is defined as

$$\text{COP} = \frac{\text{refrigeration effect}}{\text{external heat input}} = \frac{Q_E}{Q_G}$$

It is simply derived from Eq. (28) as

$$\text{COP} = \frac{(H_{10} - H_8)\,(X_4 - X_1)}{[X_1 H_5 + (X_4 - X_1)\,H_7 - X_4 H_1]} \tag{31}$$

H. Ideal Coefficient of Performance

The maximum coefficient of performance of the above absorption cycle is given by:

$$(\text{COP})_{max} = \frac{T_e(T_g - T_a)}{T_g(T_c - T_e)} \tag{32}$$

where T_e, T_a, T_c, and T_g are the absolute temperatures of the evaporator, absorber, condenser, and generator, respectively.

The ratio

$$\frac{(COP)_{actual}}{(COP)_{max}}$$

is called the 'relative performance ratio," to show the deviation from reversible cycle operation.

V. Main Components Modeling

Each of the four basic heat exchangers (condenser, absorber, generator and evaporator) is considered as a "constant temperature heat exchanger" as shown in Fig. 4. This is due to the fact that the heat transfer mechanism involves a change in phase while the temperature of one of the heat transfer fluids is kept constant.

There are two basic approaches to determine the heat transfer characteristics of each heat exchanger, namely,

(1) Using the conventional logarithmic mean temperature difference expression.

(2) Using the combined effectiveness/number of exchanger heat transfer units N_{tu}, the latter defined as (Ref. 6).

$$N_{tu} = \frac{UA}{\textit{Smaller heat capacity of the two heat transfer fluids}} \qquad (33)$$

Both approaches give a straightforward solution and there exists a one-to-one correspondence between the two sets of parameters in each case.

Because of lack of sufficient and reliable information about the overall conductance coefficient U for a LiBr-water solution, an approximate modeling trial was made by Wilber et al. (Ref. 7) in terms of a "characteristic product" (UA) associated with each heat exchanger. The latter was determined from the temperature pattern at nominal design conditions, in spite of the fact that the conductance U is a dominant function of flow rates and fluid properties for a laminar flow.

On the other hand, Lackey (Ref. 2) indicated that an increase of LiBr-water flow rate by as much as 350 percent resulted in a change of the overall temperature pattern of only 6 percent and the product (UA) was no longer considered to be a fixed property at all operating conditions. The above suggests that the main components modeling procedure would be best characterized by fixed temperature differences ΔT_i and ΔT_0, as shown in Fig. 4, rather than a fixed (UA) product. Consequently, the flow

rates of the externally heating or cooling fluids would be self-controlled to suit the variations in heat transfer rates. A good practical estimate of the inlet temperature difference ΔT_i and the outlet temperature difference ΔT_0 is 10°C and 3°C, respectively, each for the generator, absorber and condenser and 20°C and 6°C, respectively, for the air-cooled evaporator.

VI. Liquid-Liquid Heat Exchanger Modeling

This is considered as an "unbalanced counter flow heat exchanger," since the two relevant streams possess unequal heat capacities. The exact effectiveness expression (Ref. 6) as applied to LiBr-water solutions indicated that extreme off-design changes of flow rates or concentrations can change the design effectiveness value of E_L by only ± 5 percent. The consequent effect on the solution enthalpies leaving the heat exchanger, Eq. (23), is negligible. This suggests that the effectiveness E_L may be taken as a constant in the analysis without great loss of accuracy.

VII. Summary

The sequence of program calculations is summarized as follows:

Input Data

(1) t_g, °C, generator temperature

(2) t_e, °C, evaporator temperature

(3) t_c, °C, condenser temperature

(4) t_a, °C, absorber temperature

(5) E_L, exchanger effectiveness

(6) Q_E, kcal/hr, load

Steps of Analysis

(1) $X_1 = \dfrac{(49.04 + 1.125\,t_a - t_e)}{(134.65 + 0.47\,t_a)}$ kg LiBr/kg solution

(2) $X_4 = \dfrac{(49.04 + 1.125\,t_g - t_c)}{(134.65 + 0.47\,t_g)}$ kg LiBr/kg solution

* IF $0.5 < (X_1 \text{ and } X_4) < 0.65$ proceed, else stop.

(3) $H_8 = (t_c - 25)$ kcal/kg

(4) $H_{10} = (572.8 + 0.417\,t_e)$ kcal/kg

(5) $m_R = \dfrac{Q_E}{(H_{10} - H_8)}$ kg/hr

(6) $m_s = m_R \cdot \dfrac{X_4}{(X_4 - X_1)}$ kg/hr

(7) $m_w = m_R \dfrac{X_1}{X_4 - X_1}$ kg/hr

(8) $t_5 = t_g - E_L(t_g - t_a)$ °C

(9) $C_{X_1} = 1.01 - 1.23\,X_1 + 0.48\,X_1^2$ kcal/kg°C

(10) $C_{X_4} = 1.01 - 1.23\,X_4 + 0.48\,X_4^2$ kcal/kg°C

(11) $t_3 = t_a + \left[E_L \cdot \dfrac{X_1}{X_4} \cdot \dfrac{C_{x_4}}{C_{x_1}} \cdot (t_g - t_a) \right]$ °C

(12) $H_1 = (42.81 - 425.92\,X_1 + 404.67\,X_1^2)$
$\qquad + (1.01 - 1.23\,X_1 + 0.48\,X_1^2) \cdot (t_a)$ kcal/kg

(13) $H_5 = (42.81 - 425.92\,X_4 + 404.67\,X_4^2)$
$\qquad + (1.01 - 1.23\,X_4 + 0.48\,X_4^2) \cdot (t_5)$ kcal/kg

(14) $H_7 = (572.8 + 0.46\,t_g - 0.043\,t_c)$ kcal/kg

(15) $Q_C = m_R(H_7 - H_8)$ kcal/hr

(16) $Q_G = (m_w H_5 + m_R H_7 - m_s H_1)$ kcal/hr

(17) $Q_A = (m_w H_5 + m_R H_{10} - m_s H_1)$ kcal/hr

(18) $COP = Q_E/Q_G$

(19) $(COP)_{max} = \dfrac{(t_e + 273.15)\,(t_g - t_a)}{(t_g + 273.15)\,(t_c - t_e)}$

(20) relative performance ratio $= COP/(COP)_{max}$

(21) $P_e = \text{antilog} \left[7.8553 - \dfrac{1555}{t_e + 273.15} \right.$
$\left. - \dfrac{11.2414 \times 10^4}{(t_e + 273.15)^2} \right]$ mmHg

(22) $P_c = \text{antilog} \left[7.8553 - \dfrac{1555}{t_c + 273.15} \right.$
$\left. - \dfrac{11.2414 \times 10^4}{(t_c + 273.15)^2} \right]$ mmHg

VIII. Numerical Example

The following numerical example presents the performance characteristics of a typical running condition:

Input Data

$t_g = 90°C$

$t_e = 7°C$

$t_c = 40°C$

$t_a = 40°C$

$E_L = 0.8$

$Q_E = 1$ ton of refrigeration $= 3024$ kcal/hr

Cooling fluid lines to the condenser and absorber are in parallel.

Calculation Steps

$X_1 = 0.5672$, kg LiBr/kg solution

$X_4 = 0.6233$, kg LiBr/kg solution

$H_8 = 15$, kcal/kg

$H_{10} = 575.72$, kcal/kg

$m_R = 5.3931$, kg/hr

$m_s = 59.9199$, kg/hr

$m_w = 54.5268$, kg/hr

$t_5 = 50$, °C

$C_{X_1} = 0.46677$, kcal/kg°C

$C_{X_4} = 0.42982$, kcal/kg°C

$t_3 = 73.52$, °C

$H_1 = -49.9124$, kcal/kg

$H_5 = -43.9594$, kcal/kg

$H_7 = 612.48$, kcal/kg

$Q_C = 3222.3$, kcal/hr

$Q_G = 3896.9$, kcal/hr

$Q_A = 3698.6$, kcal/hr

$COP = 0.776$

$(COP)_{max} = 1.1689$

relative performance ratio $= 0.664$

$P_e = 7.45$ mmHg

$P_c = 55.37$ mmHg

The temperature profile of each heat exchanger is sketched as shown in Fig. 5. Having established an analytical procedure for the performance characteristics of a lithium bromide/water absorption system enables the study of further changes in the thermodynamic cycle. Examples of these changes are the temperature of entering heating fluid to the generator, the cooling load at the evaporator, and the temperature of entering cooling fluid at the condenser or absorber.

Definition of Symbols

A heat exchanger surface area, m²

C_x specific heat of solution of concentration X, kcal/kg solution °C

E_L effectiveness of liquid/liquid heat exchanger

H specific enthalpy, kcal/kg

m mass flow rate, kg/hr

p pressure, mm Hg

Q rate of heat transfer, kcal/hr

t temperature, °C

U overall conductance coefficient, kcal/(hr-m²-°C)

X solution concentration, kg LiBr/kg solution

Subscripts

a,A absorber

c,C condenser

e,E evaporator

g,G generator

s "strong" solution

w "weak" solution

References

1. Ellington, R. T., Kunst, G., Peck, R. E., and Reed, J. F., "The Absorption Cooling Process." Institute of Gas Technology Research Bulletin No 14, Chicago, Ill., Aug. 1957.

2. Lackey, R. S., "Solar Heating and Cooling of Buildings," Westinghouse Electric Corporation Report to the National Science Foundation, NSF-74-C584-1-2, Appendix J., May 1974.

3. Threlkeld, J. L., *Thermal Environmental Engineering*, Prentice Hall, 1970.

4. San Martin, R. L., and Couch, W. A., "Modeling of Solar Absorption Air Conditioning," Institute of Environmental Science Proceedings, Vol. 1, 1975, pp. 186–189.

5. Pennington, W., "How to Find Accurate Vapor Pressures of LiBr-Water Solutions," *Refrigeration Engineering*, May 1955.

6. Kays, W. M., and London, A. L., *Compact Heat Exchangers*, McGraw Hill Book Co., Inc., 1958.

7. Wilbur, P. J., and Mitchell, C. E., "Solar Absorption Air Conditioning Alternatives," *Solar Energy*, Vol. 17, 1975, pp. 193–199.
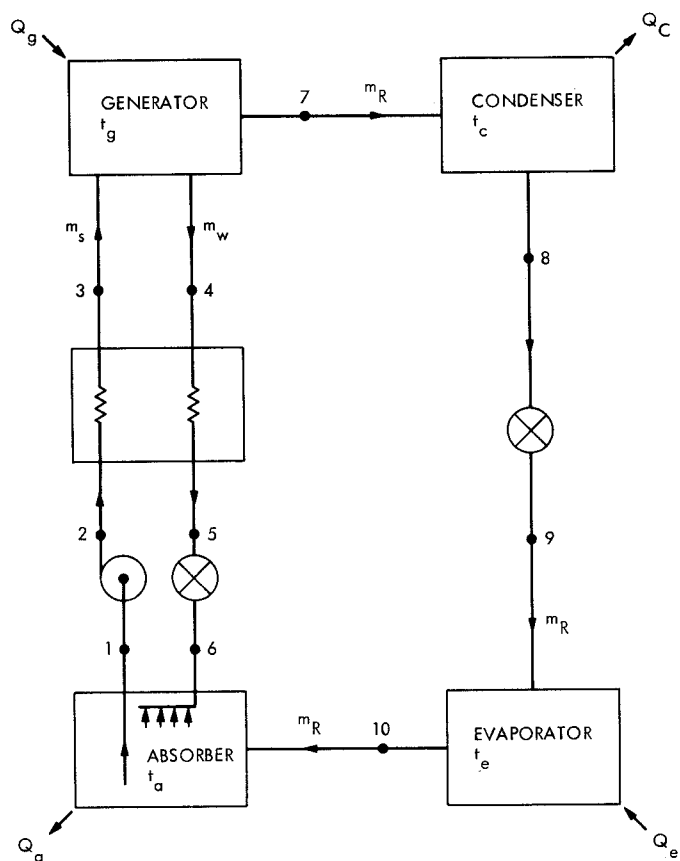
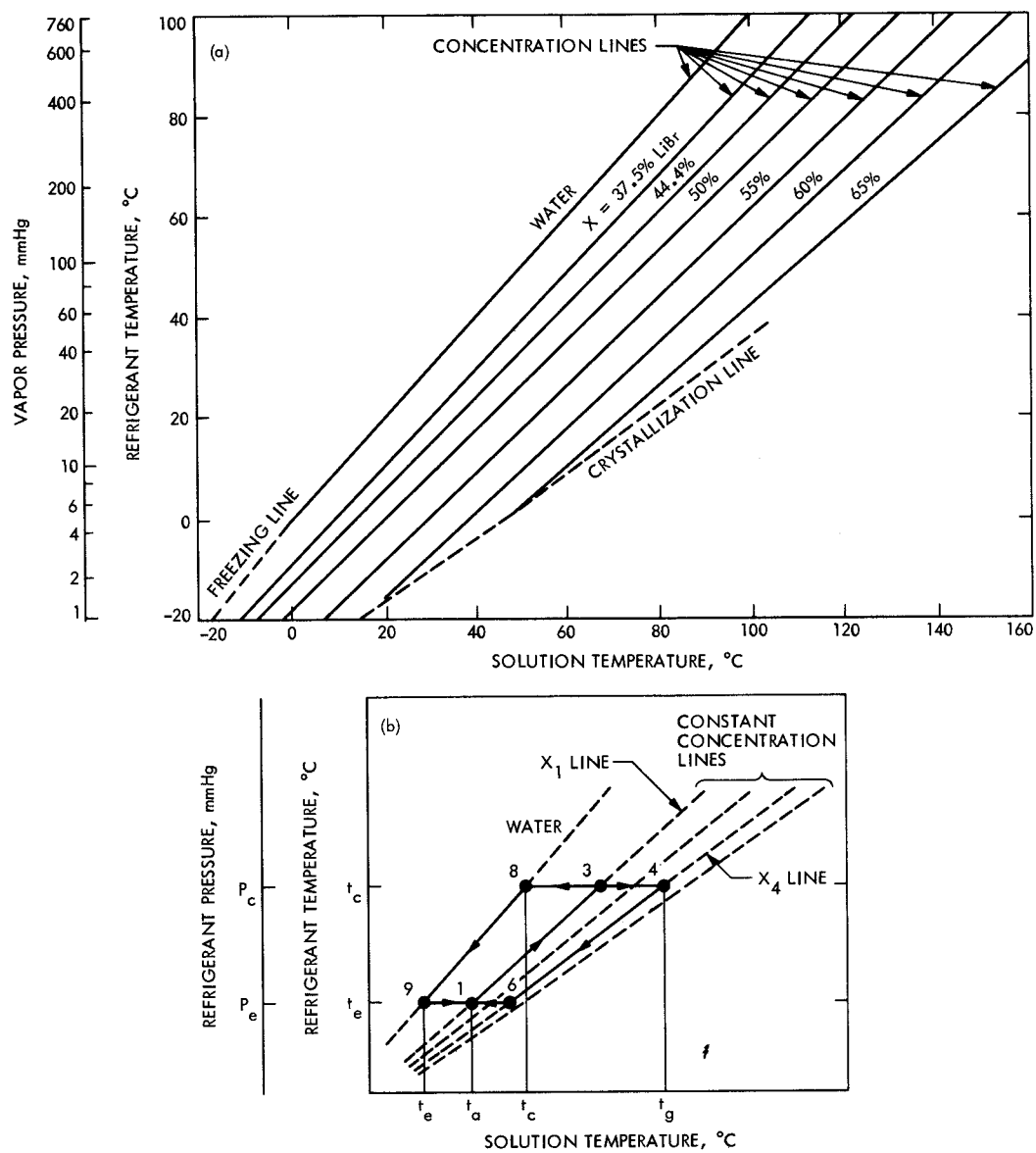Fig. 1. Flow diagram for a lithium bromide/water
absorption system

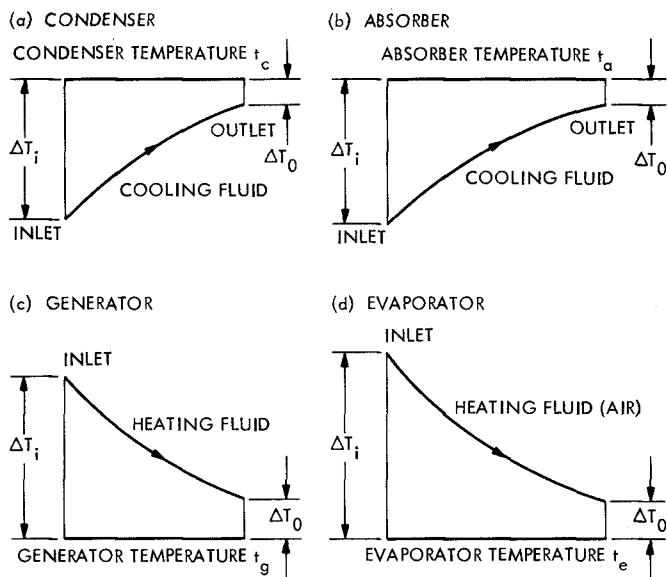Fig. 2. Equilibrium chart for lithium bromide/water solution

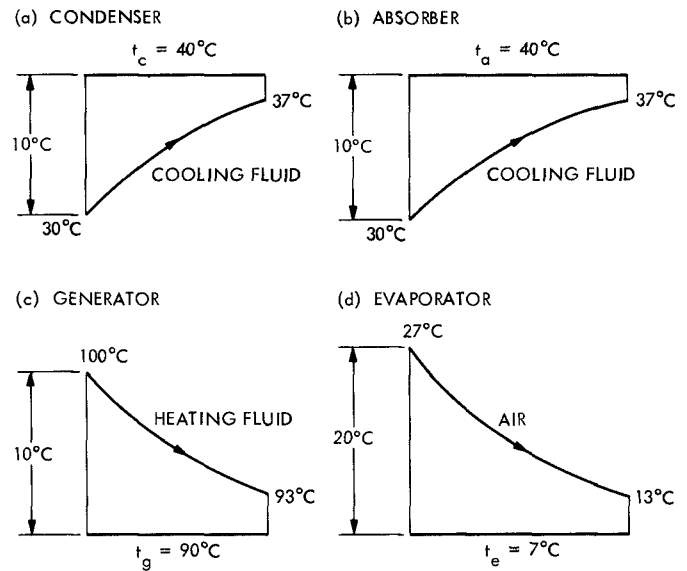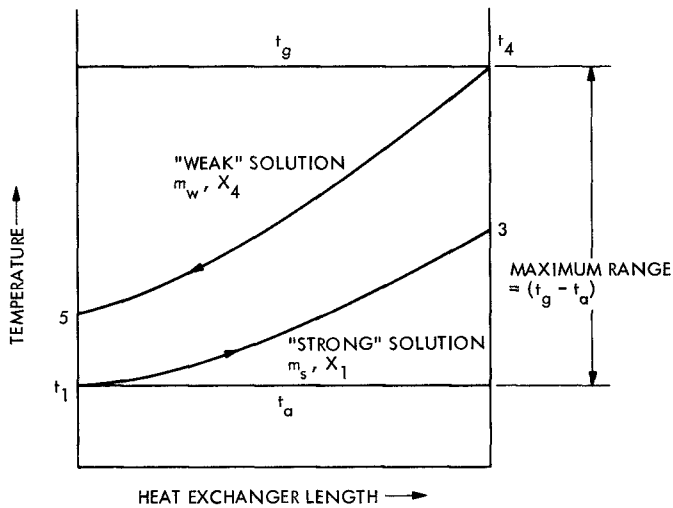Fig. 3. Effectiveness of liquid-liquid heat exchanger

Fig. 4. Temperature profiles along the four main components: condenser, absorber, generator and evaporator
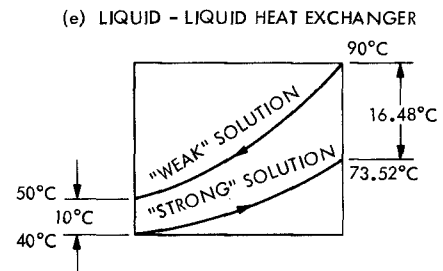

Fig. 5. Temperature patterns along the heat exchangers for the numerical example